Advance Noise Control Fan II Test Rig Fan Support Bearings Trade Study

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Background Information

- Since 1995 the Advanced Noise Control Fan (ANCF) has significantly contributed to the advancement of the understanding of the physics of fan ***tonal*** noise generation.

• The 9’x15’ WT has successfully tested multiple high speed fan designs over the last several decades.

•This advanced several tone noise reduction concepts to higher TRL and the validation of fan tone noise prediction codes

Current GRC Facilities

Capabilities of current GRC Fan Noise Test Facilities

- ANCF @ AAPL (TRL 2-3) :

  Low speed / ultra-low pressure rise / unique acoustic measurements

  / limited aero measurements / high flexibility / parametric studies

  / low cost

- UHB @ 9x15 LSWT (TRL 4-5):

  High speed / pressure rise / aero & performance measurements / acoustic measurements w caveats / forward flight effects / point design / high cost

- W8 (TRL 4):

  High speed / pressure rise / aero & performance measurements / moderate costs

Background Information

NEED:

A new Fan Test Rig to bridge from TRL 3 to 5 enabling the successful completion of NASA/Industry noise reduction program goals.

Concept Study Assumptions

What would it look like?

(High level design requirements)

•All electric drive to minimize external support ($) (consider alternatives)
- Minimize component noise level (initial metric > 20? dB below WT)

• Tested designs transferable to 9x15 WT - 22” fan diameter*
- (suggested actual hardware a plus)

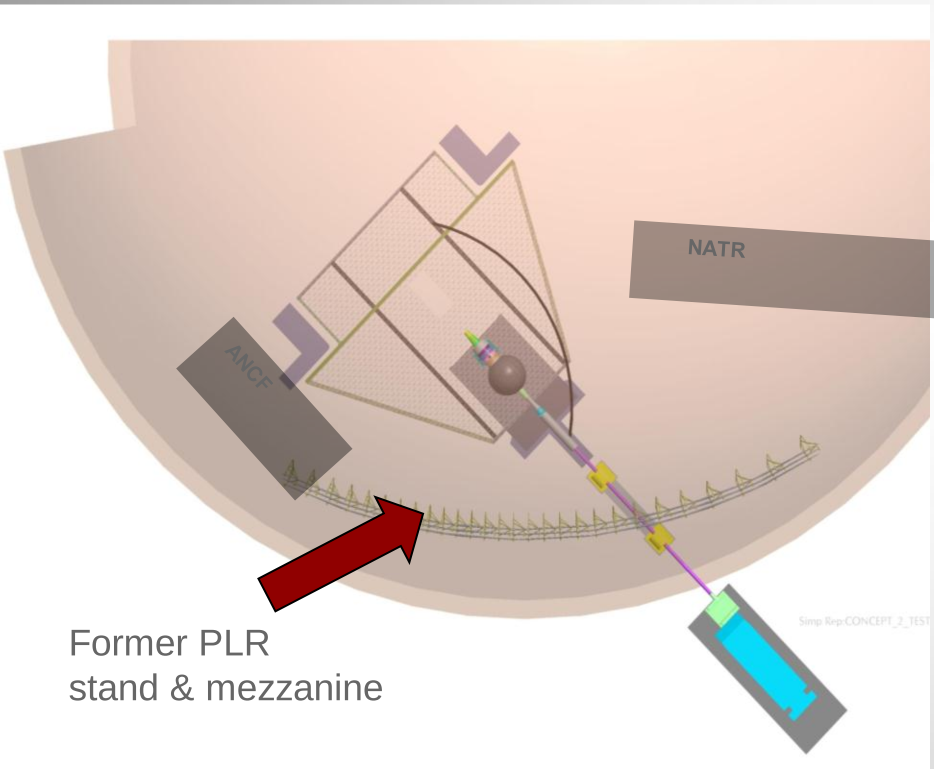
• Maintain current measurement capabilities.
- Far field, in-duct, wall pressures, flow diagnostics, aero-performance

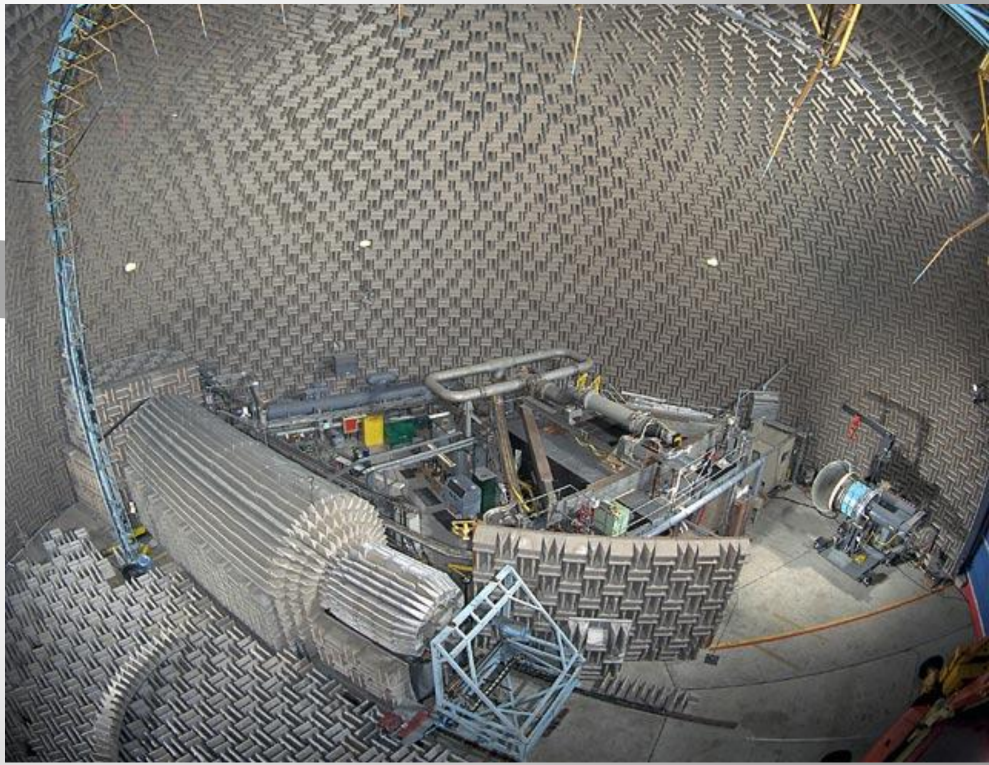
• Sited in AAPL - Minimal impact on existing rigs
- Ambient temperature conditions

• Static - no external flow lines to complicate / no forward flight effects

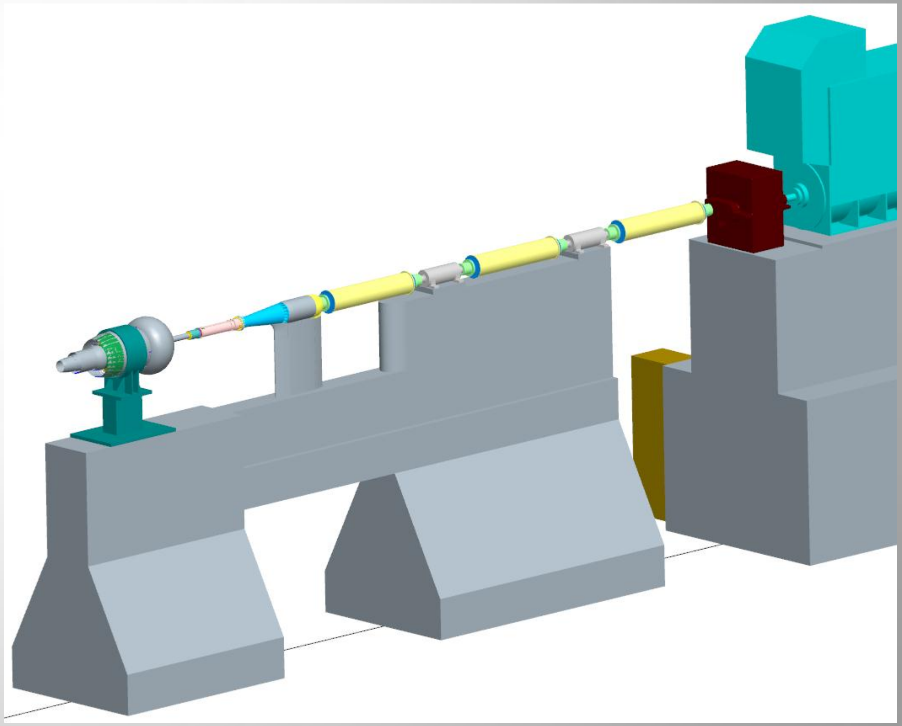
ANCF II Location in AAPL

Proposed location of the new test rig with respect to current facility layout.

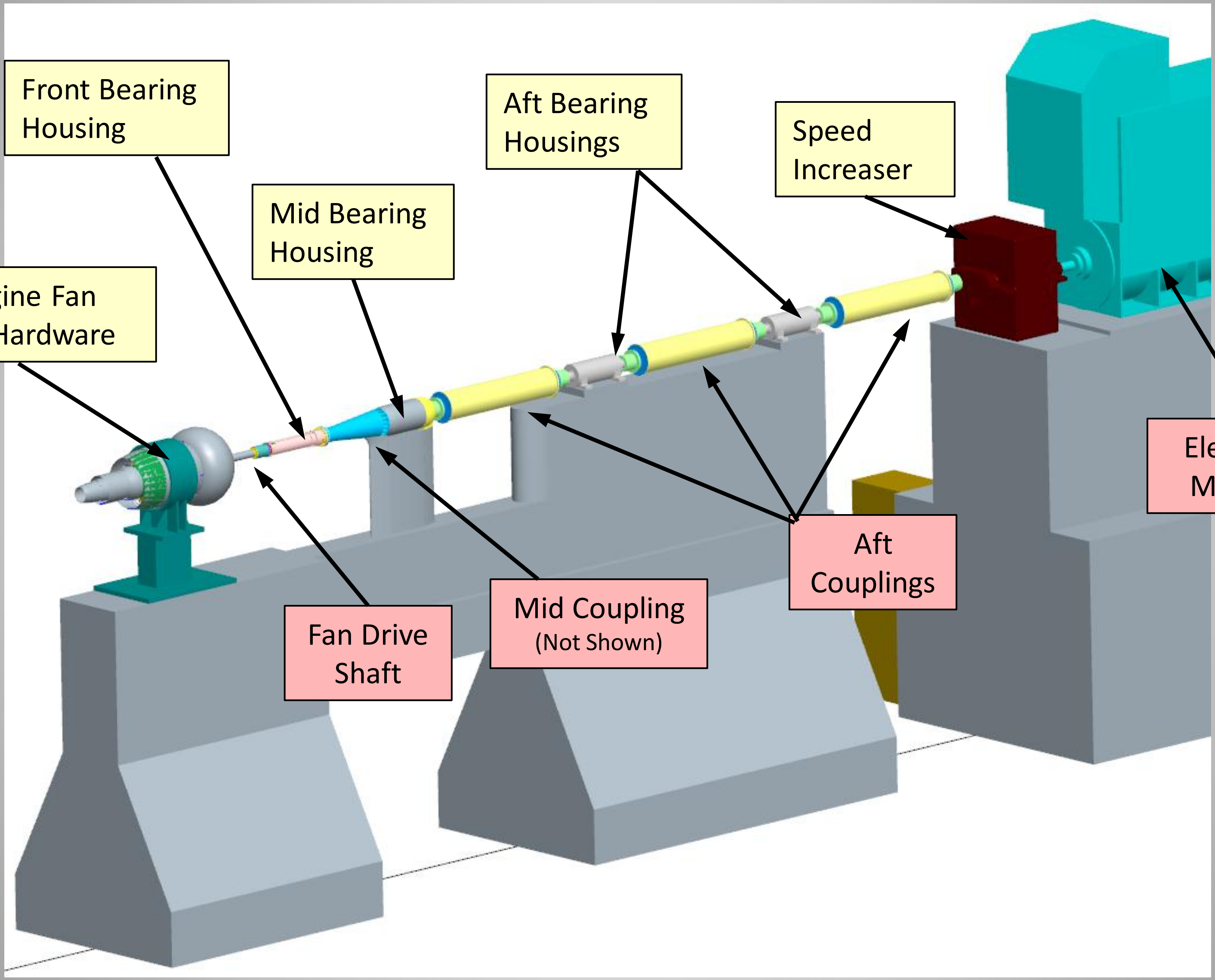




Background Information

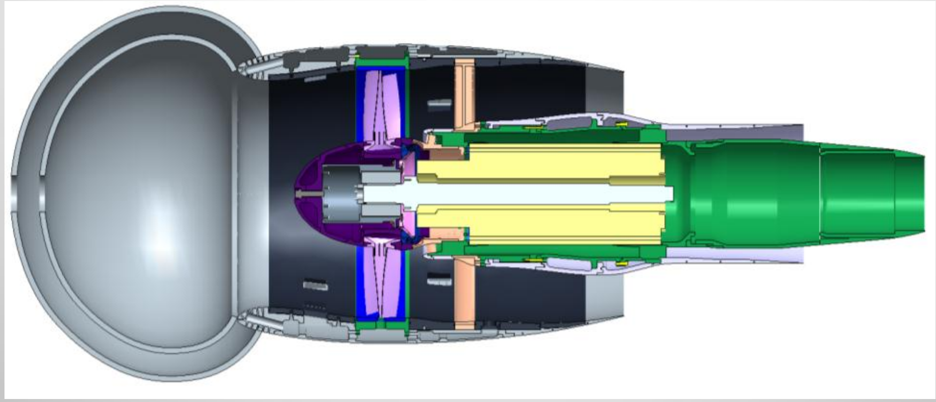


Front Driven Fan- Test Rig Overview



Objective: Bearings for the inner fan shaft of the ANCFII test rig needed to be sized and selected.

- (I) **Background**
  (rationale / current facilities)
- (I) **Summary of Preliminary Feasibility Study**
  (drive concepts / facility requirements)
- (II) **Recommendation**



Summary of Information

- Determined early grease pack bearings cannot carry the maximum axial load (2,000 lbs) at the maximum speed (15,000 rpm) without overheating.
- Six feasible core bearing technologies which formed eight different concepts were identified as feasible for this application.
- The concepts were screened down using the “Pairwise Comparison Method” to three that were then further developed in parallel until it was clear what the best option was.

Identified Technologies	
1	Oil Angular Contact
2	Grease Pack Angular Contact (only for radial load support)
3	Hydrostatic Air Bearings
4	Hydrodynamic Air Bearings
5	Magnetic Bearings
6	Electromagnets

Identified Feasible Technologies

Formed Concepts

Option	Radial Thrust Component	Axial Thrust Component
1	Oil Angular Contact	Oil Angular Contact
2	Hydrostatic Porous Media	Hydrostatic Air Thrust Pad
3	Hydrostatic Porous Media	Standard Electromagnet
4	Hydrostatic Porous Media	Hydrodynamic Foil Thrust Bearing
5	Hydrodynamic Foil Bearing	Standard Electromagnet
6	Hydrodynamic Foil Bearing	Hydrodynamic Foil Thrust Bearing
7	Grease Angular Contact	Standard Electromagnet
8	Magnetic Radial Bearing	Magnetic Thrust Bearing

Screening Criteria

- ***Performability -*** The concepts ability to minimize any negative effects on the acoustic environment caused by the bearing or any supporting hardware, wiring, or hoses. The ability for the bearings to go above and beyond the minimum requirements.

- ***Durability -*** The ability to tolerate excessive (unplanned) loads, vibrations, and misalignments without degrading performance and the ability to minimize the propagation of failures.

- ***Affordability -*** The ability to minimize initial and life cycle costs

- ***Risk -*** Any technology that NASA has not operated in a similar application is considered a higher risk technology. In addition, any technology that would require prototype testing would fall into a higher risk category.

- ***Usability/Operability/Maintainability -*** Minimize complexity of operation, time required for setup or preparation for testing, and down-time between testing. The concept will work when you need it to without any un-necessary realignments, adjustments, etc.

- ***Acceptability –*** Technical acceptance by the customer. Minimize end user resistance to implementation and use.

Concept Study Assumptions

- All the concepts shown have been put together with the assumption that they will all be able to feasibly perform the minimum requirements of the test rig.
- There is a feasible way to get oil, air, or wires through the stators without the addition of extra service struts or obstructions to the acoustic environment.
- No service lines would have to be added for the hydrodynamic bearings.
  - This turned out to be an incorrect assumption because we later found out that they do require a low pressure feed line.
- The grease pack bearings chosen for the application would not overheat even though they are being operated over experts in the field recommend as the speed limits (200,000 dN)
  - This value is lower than the value the manufacturers advertise as being permissible though.
- The hydrostatic and hydrodynamic bearings would not have to be changed out in less than a 10 year lifespan.
- That the hydrostatic and hydrodynamic bearings would be the easiest bearings to use and operate. They would require less maintenance than any other bearing choice.
- Oil AC bearings would require the most maintenance of all the concepts and require the most monitoring during operation.
- We did not weight the concepts on acceptability at this time because, one, it has the least weight of any of the screening criteria, and the top concepts would not have been affected if they had been weighed.

Screening Criteria Weighting

	Performability	Affordability	Implementation Risk	Useability/ Operability/ Maintainability	Durability (Robustness)	Acceptability	Row Sum	%
Performability		10	5	1	5	10	31	33
Affordability	0.1		0.2	0.2	0.2	5	5.7	6
Implementation Risk	0.2	5		5	0.2	5	15.4	16
Useability/ Operability/ Maintainability	1	5	0.2		0.2	10	16.4	17
Durability (Robustness)	0.2	5	5	5		10	25.2	27
Acceptability	0.1	0.2	0.2	0.1	0.1		0.7	1
						Total =	94.4	100

- The "Pairwise Comparison Method" requires that first the screening criteria are weighted and then the concepts are weighted for each screening criteria using the same matrix style.

Performability	Oil AC/ Oil AC	HydroStat PM/ Hydrostat PM	HydroStat PM/ Electromagnet	HydroStat PM/ Hydrodyn FB	Hydrodyn FB/ Electromagnet	Hydrodyn FB/ Hydrodyn FB	Grease AC/ Electromagnet	Row Sum	%
Oil AC/ Oil AC		0.1	0.2	0.1	0.2	0.1	0.2	0.9	0.6
HydroStat PM/ Hydrostat PM	10		5	0.2	5	0.1	5	25.3	18.3
HydroStat PM/ Electromagnet	5	0.2		0.2	0.2	0.1	5	10.7	7.7
HydroStat PM/ Hydrodyn FB	10	5	5		5	0.2	5	30.2	21.8
Hydrodyn FB/ Electromagnet	5	0.2	5	0.2		0.2	5	15.6	11.3
Hydrodyn FB/ Hydrodyn FB	10	10	10	5	5		10	50	36.1
Grease AC/ Electromagnet	5	0.2	0.2	0.2	0.2	0.1		5.9	4.3
							Total =	138.6	100

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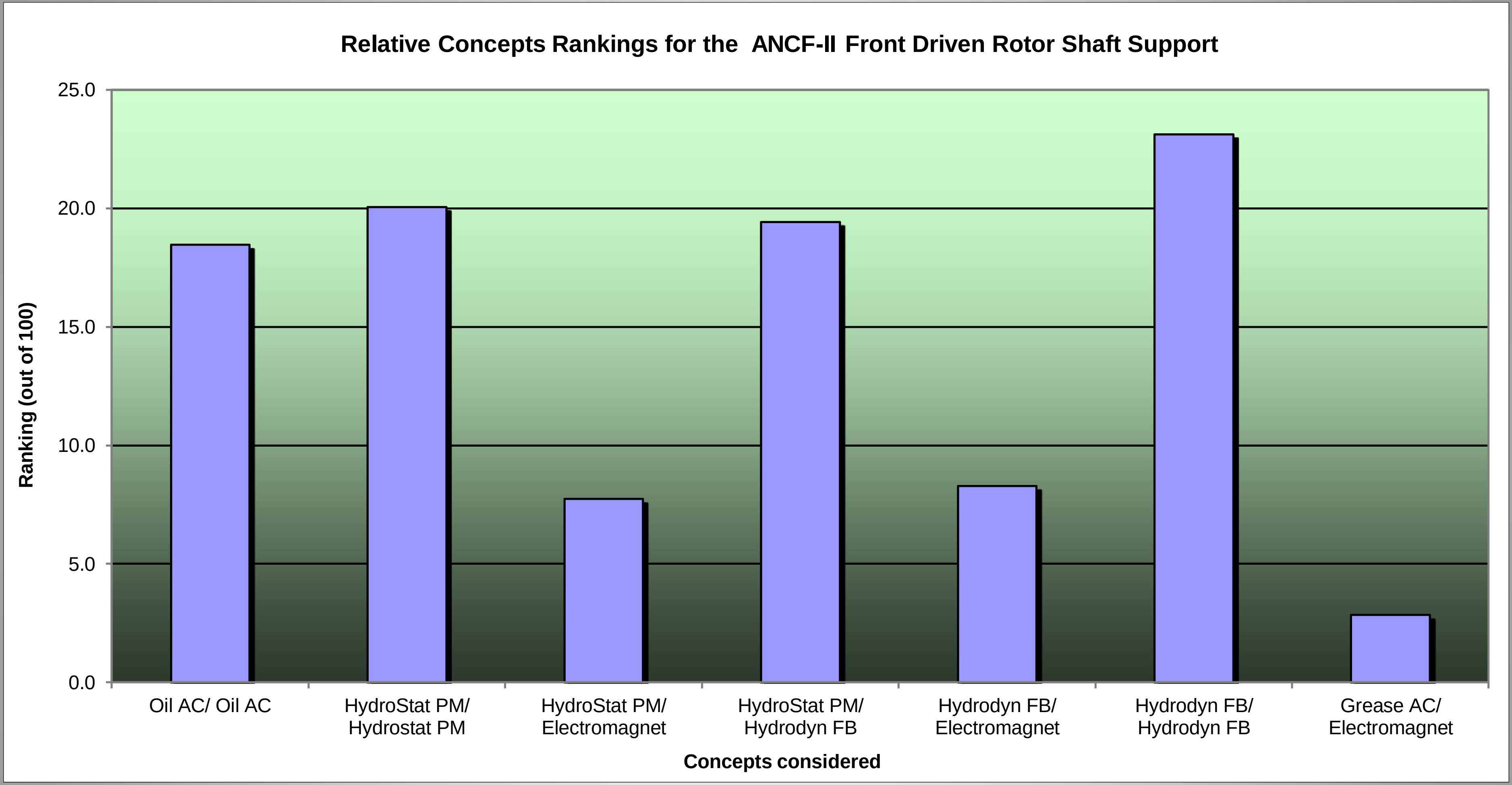
Implementation Risk	Oil AC/ Oil AC	HydroStat PM/ Hydrostat PM	HydroStat PM/ Electromagnet	HydroStat PM/ Hydrodyn FB	Hydrodyn FB/ Electromagnet	Hydrodyn FB/ Hydrodyn FB	Grease AC/ Electromagnet	Row Sum	%
Oil AC/ Oil AC		5	10	10	10	5	10	50	45.6
HydroStat PM/ Hydrostat PM	0.2		5	1	5	1	5	17.2	15.7
HydroStat PM/ Electromagnet	0.1	0.2		0.2	1	0.2	1	2.7	2.5
HydroStat PM/ Hydrodyn FB	0.1	1	5		5	1	5	17.1	15.6
Hydrodyn FB/ Electromagnet	0.1	0.2	1	0.2		0.2	1	2.7	2.5
Hydrodyn FB/ Hydrodyn FB	0.2	1	5	1	5		5	17.2	15.7
Grease AC/ Electromagnet	0.1	0.2	1	0.2	1	0.2		2.7	2.5
							Total =	109.6	100

Useability/ Operability/ Maintainability	Oil AC/ Oil AC	HydroStat PM/ Hydrostat PM	HydroStat PM/ Electromagnet	HydroStat PM/ Hydrodyn FB	Hydrodyn FB/ Electromagnet	Hydrodyn FB/ Hydrodyn FB	Grease AC/ Electromagnet	Row Sum	%
Oil AC/ Oil AC		0.1	0.2	0.2	0.2	0.2	1	1.9	1.9
HydroStat PM/ Hydrostat PM	10		5	1	5	1	5	27	27.5
HydroStat PM/ Electromagnet	5	0.2		0.2	1	0.2	5	11.6	11.8
HydroStat PM/ Hydrodyn FB	5	1	5		5	1	5	22	22.4
Hydrodyn FB/ Electromagnet	5	0.2	1	0.2		0.2	5	11.6	11.8
Hydrodyn FB/ Hydrodyn FB	5	1	5	1	5		5	22	22.4
Grease AC/ Electromagnet	1	0.2	0.2	0.2	0.2	0.2		2	2.0
							Total =	98.1	100

Durability (Robustness)	Oil AC/ Oil AC	HydroStat PM/ Hydrostat PM	HydroStat PM/ Electromagnet	HydroStat PM/ Hydrodyn FB	Hydrodyn FB/ Electromagnet	Hydrodyn FB/ Hydrodyn FB	Grease AC/ Electromagnet	Row Sum	%
Oil AC/ Oil AC		5	5	5	5	5	10	35	34.6
HydroStat PM/ Hydrostat PM	0.2		5	1	5	1	5	17.2	17.0
HydroStat PM/ Electromagnet	0.2	0.2		0.2	1	0.2	5	6.8	6.7
HydroStat PM/ Hydrodyn FB	0.2	1	5		5	1	5	17.2	17.0
Hydrodyn FB/ Electromagnet	0.2	0.2	1	0.2		0.2	5	6.8	6.7
Hydrodyn FB/ Hydrodyn FB	0.2	1	5	1	5		5	17.2	17.0
Grease AC/ Electromagnet	0.1	0.2	0.2	0.2	0.2	0.2		1.1	1.1
							Total =	101.3	100

OVERALL WEIGHT	Performability	Affordability	Implementation Risk	Useability/ Operability/ Maintainability	Durability (Robustness)	Acceptability	Row Sum	%
Oil AC/ Oil AC	0.0021	0.0114	0.0744	0.0034	0.0922	0.0011	0.1846	18.5
HydroStat PM/ Hydrostat PM	0.0599	0.0209	0.0256	0.0478	0.0453	0.0011	0.2006	20.1
HydroStat PM/ Electromagnet	0.0254	0.0086	0.0040	0.0205	0.0179	0.0011	0.0775	7.8
HydroStat PM/ Hydrodyn FB	0.0716	0.0120	0.0255	0.0390	0.0453	0.0011	0.1943	19.4
Hydrodyn FB/ Electromagnet	0.0370	0.0025	0.0040	0.0205	0.0179	0.0011	0.0830	8.3
Hydrodyn FB/ Hydrodyn FB	0.1185	0.0019	0.0256	0.0390	0.0453	0.0011	0.2314	23.1
Grease AC/ Electromagnet	0.0140	0.0031	0.0040	0.0035	0.0029	0.0011	0.0286	2.9
						Total =	1.0	100

Conceptual Screening Results



Magnetic bearings were not considered because they were found to be an order of magnitude higher in cost than the other options and the companies we talked to were not sure they could develop a bearing to meet the requirements.

Conceptual Screening Results

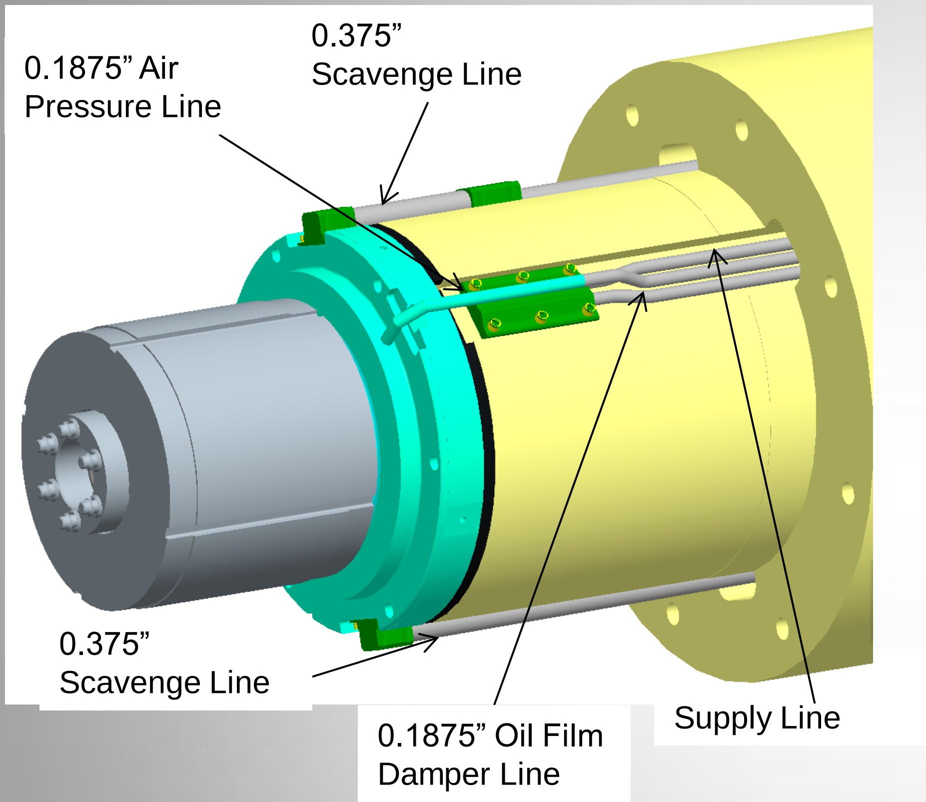
Option	Radial Thrust Component	Axial Thrust Component
1	Oil Angular Contact	Oil Angular Contact
2	Hydrostatic Porous Media	Hydrostatic Air Thrust Pad
3	Hydrodynamic Foil Bearing	Hydrodynamic Foil Thrust Bearing

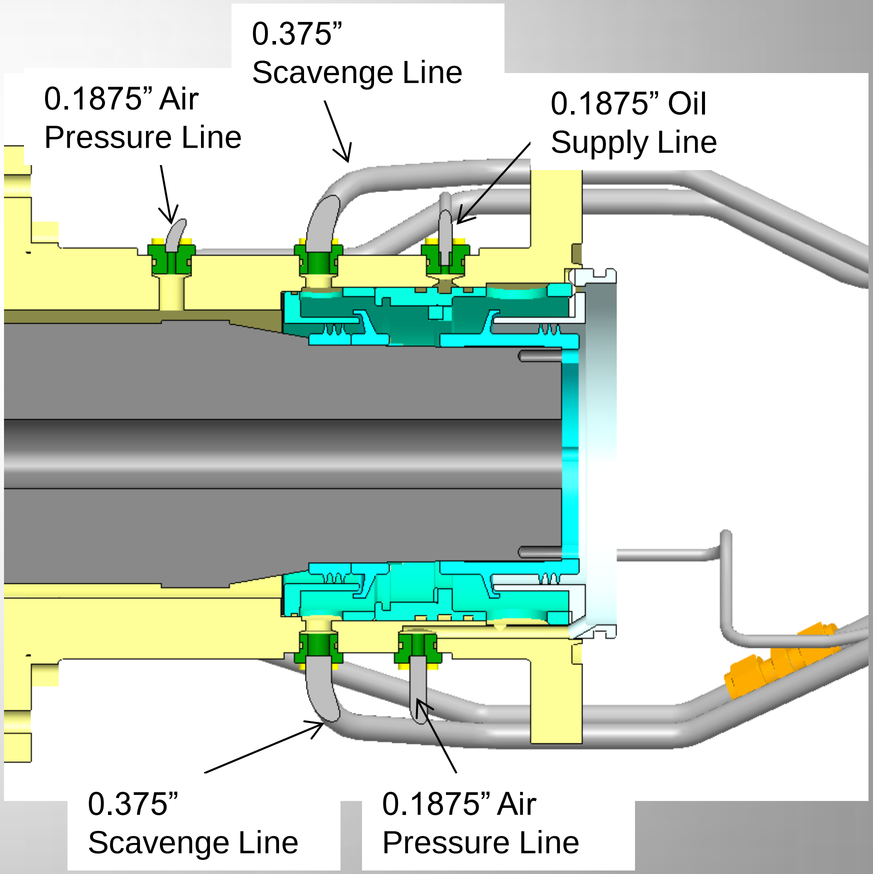
- Based on the results from the study, it was determined that three different technologies were clearly superior to the others.

- From this point, the remaining questions need to be addressed for each concept
  - Is it possible for each lubrication method required to go through the stators?
  - Could another method of be used to get the oil or air into the bearings?
  - How much would each system cost over a period of ten years?
  - Are there any newly identified “show stoppers” as each option was developed further?

Oil Lubrication Supply Line Information

Line Purpose	*Diameter and Quantity*	*Total Area*
Oil Supply Lines	0.1875" x 3	0.09 in^2
Oil Scavenge Lines	0.375" x 4	0.44 in^2
Air Pressure Lines	0.1875" x 3	0.09 in^2
	Sum:	0.62 in^2



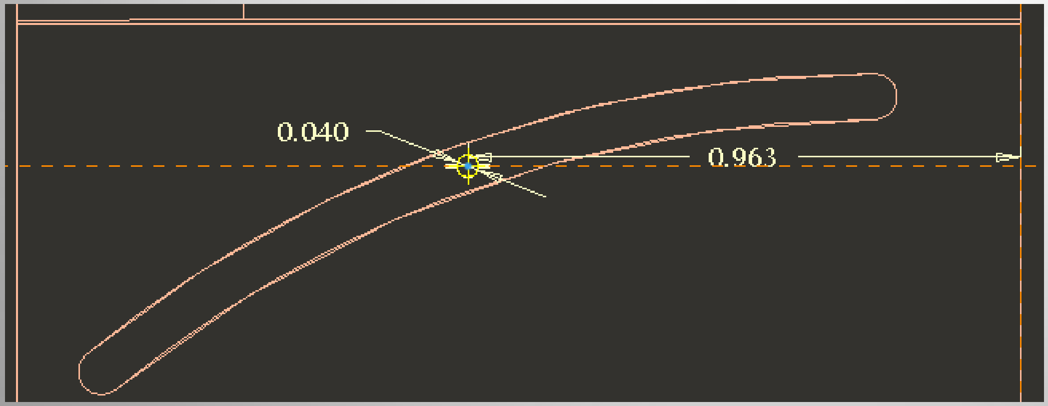


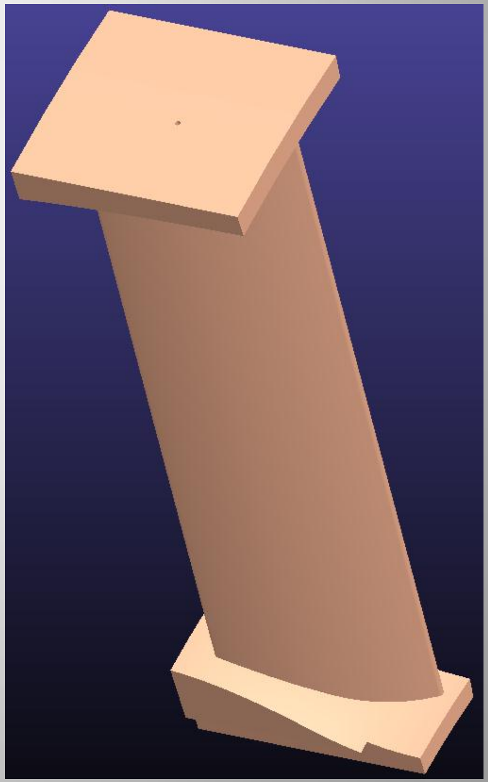
Assumed we would need to match the total supply line area that the UHB rig was currently operating with successfully

Holes Through Stators

- The current stators modeled in CAD have a maximum thickness of 0.08”.
- Assuming the smallest wall thickness allowed is 0.02”, the largest hole size is 0.04” in diameter.
- There are 45 stators modeled in the current CAD model.
- Even with 4 holes per stator, its NOT possible to get the needed supply area.

1 Hole per Stator	0.0565 in^2
2 Holes per Stator	0.113 in^2
3 Holes per Stator	0.170 in^2
4 Holes per Stator	0.226 in^2



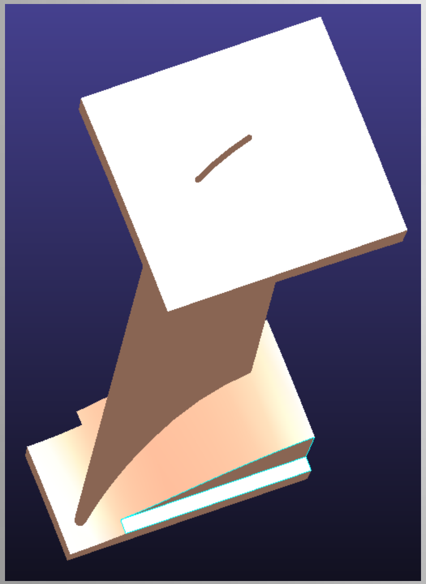


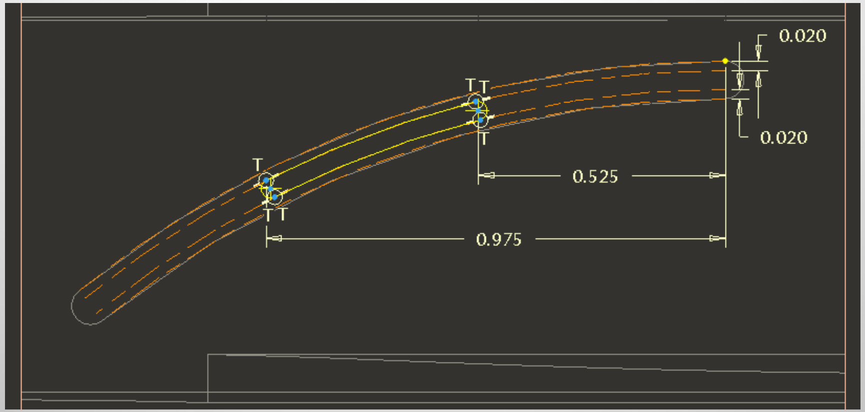
Slots Through Stators

- Instead of using holes, the possibility of slots was conceptualized.
- First using a slot that went from 35% to 65% of the stator chord length was used and then the feasibly biggest slot possible.
- Having slots in the stators could be a possible way of getting the required supply quantities, except manufacturing stators with slots drives the cost of manufacturing up to extremely expensive levels.

1 Slot per Stator	0.9045 in^2
Slots in 31 Stators	0.62 in^2

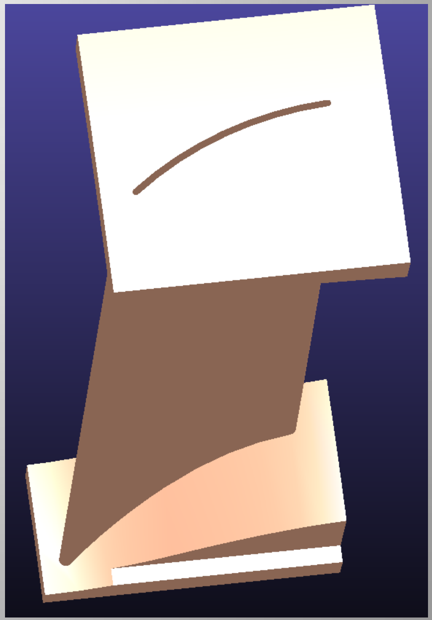
Case 1: 35% to 65% Chord Length Slot





1 Slot per Stator	2.3625 in^2
Slots in 12 Stators	0.63 in^2

Case 2: Feasibly Biggest Slot



Air Supply Lines

Assumptions and Methodology

- Utilized two different methods (Chen and Churchill) of determining the pressure drop in different sizes of constant diameter ducts.
- Both methods assume that the flow is going through a plastic tube
- Assume the pressure supplied to the bearings does not drop below 100 psi on any bearing to be conservative
- Taken from piping manufacturers, usually a pressure drop over 10% is undesirable. I assumed 5% to be conservative and allow for future unknown losses.
- Assume the diameter would only have to be at a minimum through the stator section (approx. 1 ft) and otherwise could expand to a larger diameter to decrease the pressure drop.
- Did not take into account the loss effects due to connectors and the supply piping needed to reach the test rig
- Value for density and the dynamic viscosity were both taken at standard temperature and pressure.
- The air supply source has not been finalized, but once it has, the acceptable pressure drop could become less stringent.

pipe length	L	1	ft
Density of Air	ρ	2.38E-03	$slug/ft^3$
Dynamic Viscosity of Air	μ	4.01E-07	$lbf\text{-}s/ft^2$
Pipe Roughness	ε	5.00E-06	ft

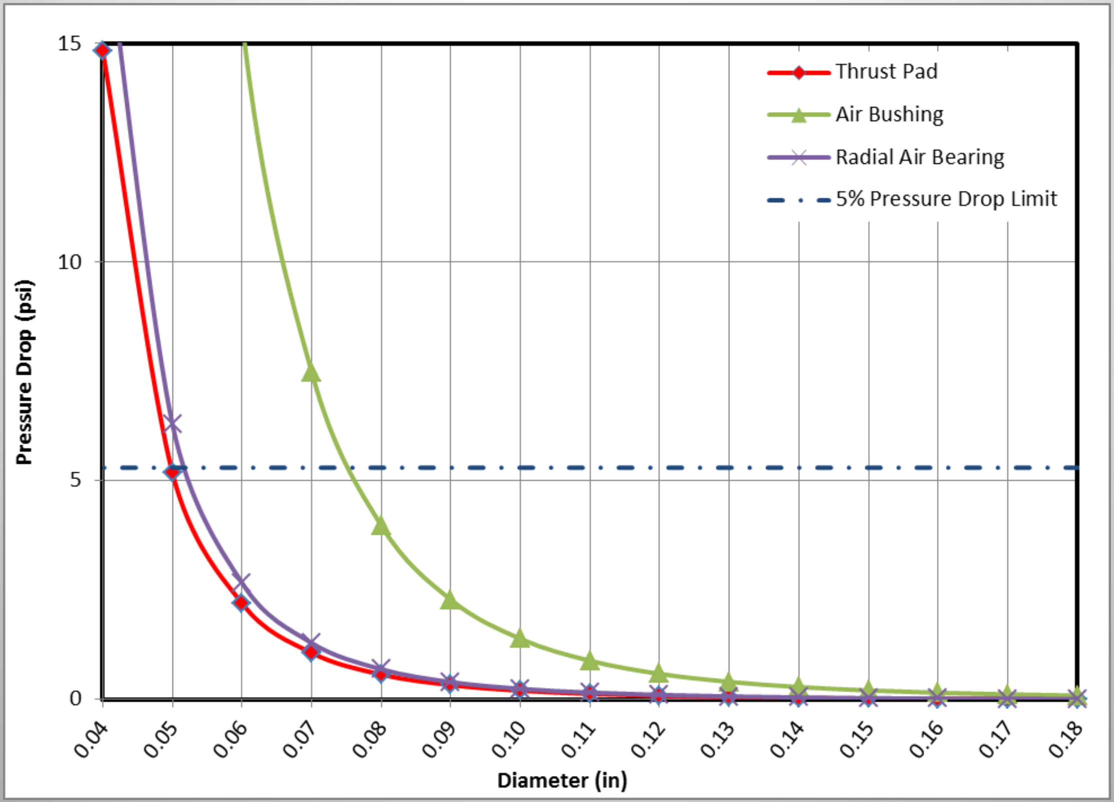
Pressure Supplied by Source	106	psi
5% Pressure Drop	5.3	psi
Minimum Operating Pressure	100.7	psi

Bearing Type	Flow Rate Required (ft^3/min)
Radial Air Bearing	0.25
Air Bushing	0.70
Air Thrust Pad	0.23

Air Supply Line Sizing

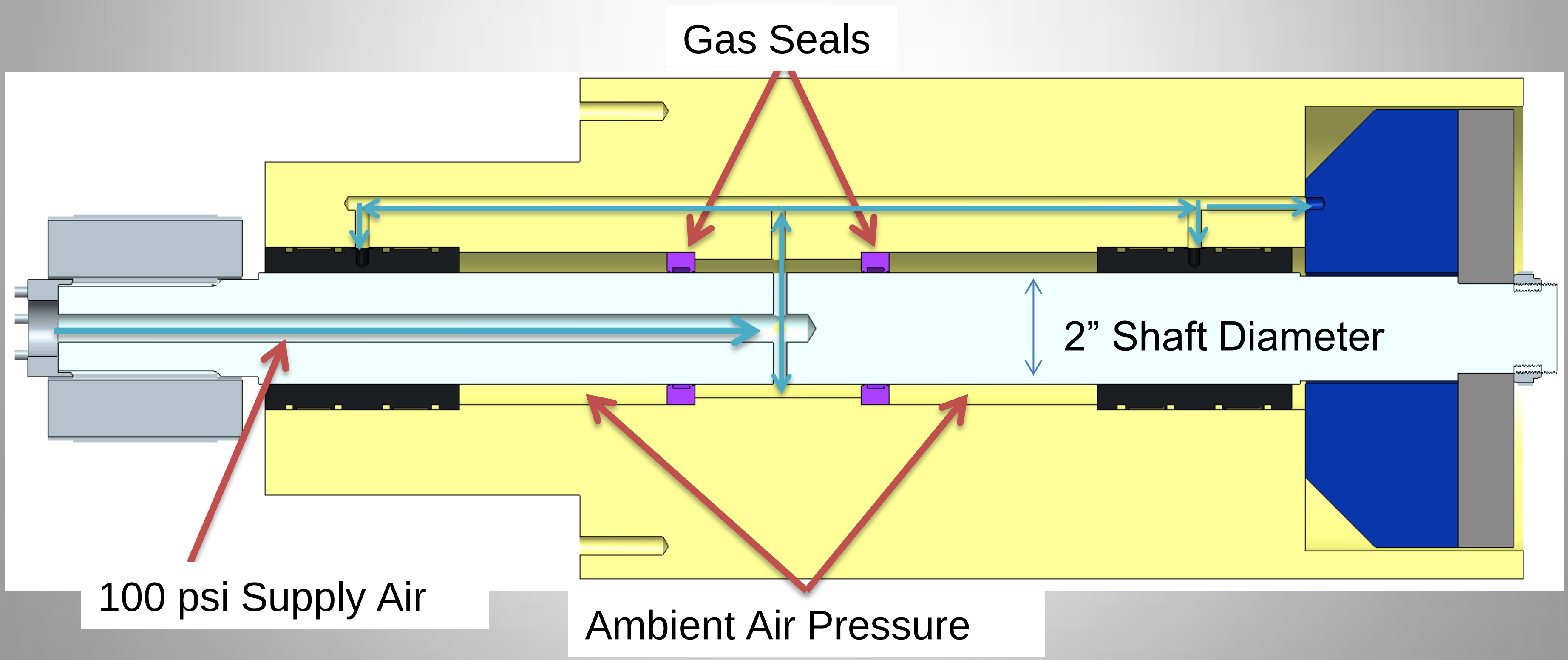
- The diameter through the stator section should not be smaller than 0.055" for the thrust pad and radial air bearing. The air bushing could not be smaller than 0.075" in diameter.

- Based on this, it is definitely not more feasible to get air supply through the stators over oil supply. Both would still need slots which are expensive to manufacture.



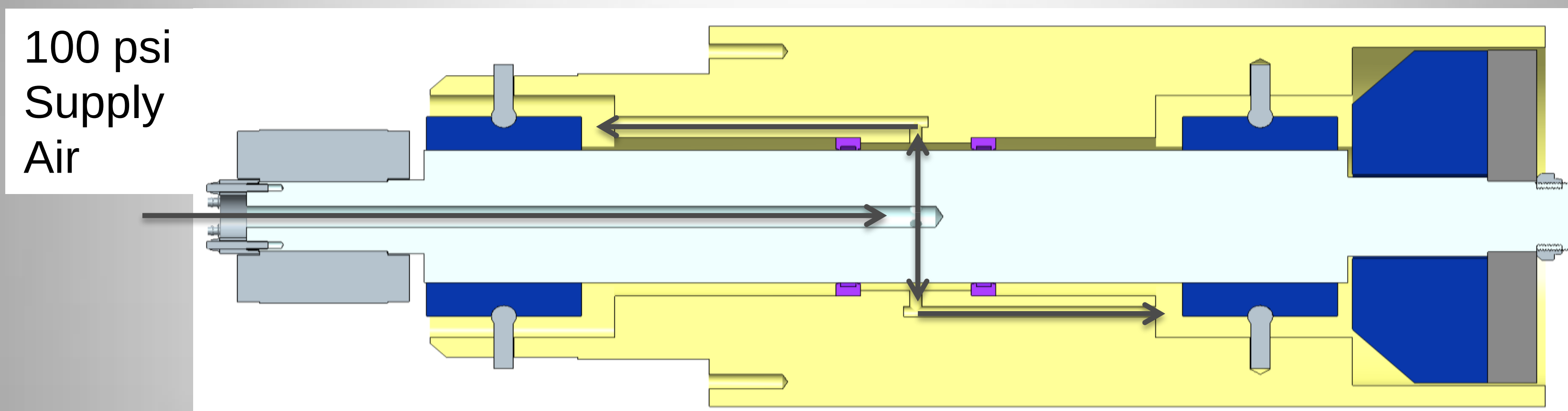
Different Supply Method – Air Only

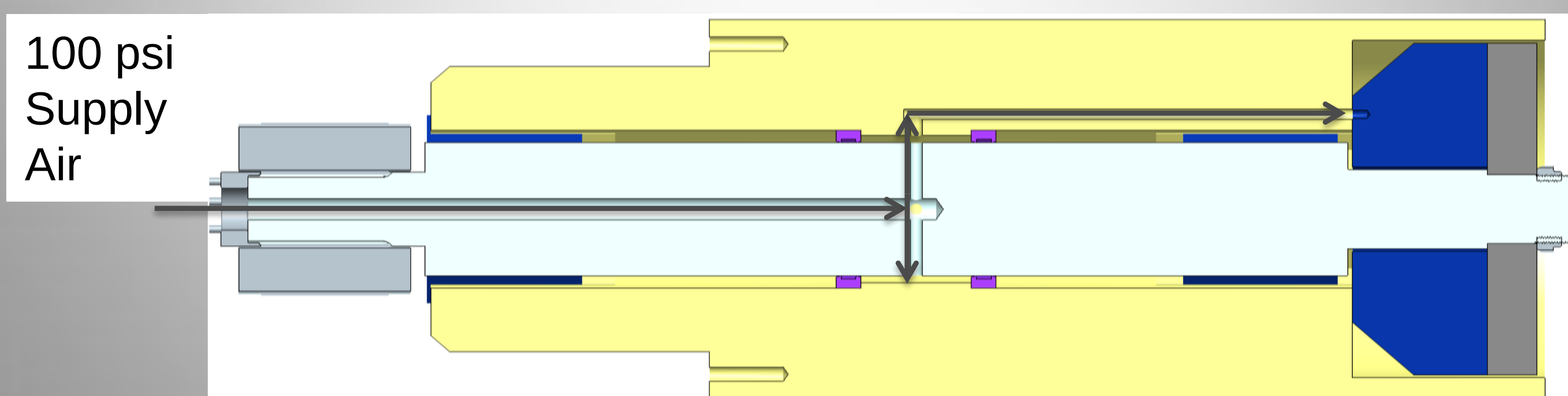
- Supply the high pressure air without enforcing design changes to the stators.
- Would allow for larger diameter feed lines with lower pressure drops.



Different Supply Method – Air Only

- This method of air supply would require gas seals on both the entrance and the exits.
- Air would be input through the main support structure in front of the test section and then fed through all the hollow shafting until it reached the middle of the inner fan shaft where it would then be guided into air ducts that lead to each bearing.





Option 1 Cost Estimate

Oil Rolling Element Bearings						
Description	Part #	Cost		Quantity		Total
GE Split Race Radial Ball Bearing	4013428-062	$1,100.00	per bearing	2	bearings	$ 2,200.00
GE Roller Bearing	5014T13P06	$1,100.00	per bearing	2	bearings	$ 2,200.00
Manufacturing Cost of Required Oil Supply Hardware, Oil Scavenge Hardware, Seals	5923-303-1	$ 105.00	per hour	50	hours	$ 5,250.00
	5923-303-2	$ 105.00	per hour	50	hours	$ 5,250.00
	5923-304-1	$ 105.00	per hour	50	hours	$ 5,250.00
	5923-305-1	$ 105.00	per hour	50	hours	$ 5,250.00
	5923-306-1	$ 105.00	per hour	50	hours	$ 5,250.00
	5923-307-1	$ 105.00	per hour	50	hours	$ 5,250.00
	5923-308-1	$ 105.00	per hour	50	hours	$ 5,250.00
	5923-309-1	$ 105.00	per hour	50	hours	$ 5,250.00
	5923-309-2	$ 105.00	per hour	50	hours	$ 5,250.00
	5923-310-1	$ 105.00	per hour	50	hours	$ 5,250.00
	5923-311-1	$ 105.00	per hour	50	hours	$ 5,250.00
	5923-311-2	$ 105.00	per hour	50	hours	$ 5,250.00
	5923-312-1	$ 105.00	per hour	50	hours	$ 5,250.00
	5923-313-1	$ 105.00	per hour	50	hours	$ 5,250.00
Oil Supply Pump	N/A	$2,000.00	per unit	1	units	$ 2,000.00
Oil Reservoir	N/A	$ 500.00	per unit	1	units	$ 500.00
Supply Oil	N/A	$ 40.00	per gallon	110	gallons	$ 4,400.00
Oil Filters	N/A	$ 75.00	per unit	20	units	$ 1,500.00
2 Stage Air Filter	S90L006	$ 225.00	per unit	1	units	$ 225.00
Air Dryer	S90L007	$ 150.00	per unit	1	units	$ 150.00
Air Regulator	S90R002	$ 90.00	per unit	1	units	$ 90.00
Replacement Filter	S90L008	$ 150.00	2 per unit	20	units	$ 3,000.00
Replacement Desiccant Dryer	S90L009	$ 30.00	per unit	20	units	$ 600.00
1/4" NPT Hex Nipple	S90F091	$ 12.00	per unit	2	units	$ 24.00
100 ft - 0.25" Plastic Air Tubing	S90T004	$ 1.00	per ft	50	ft	$ 50.00
100 ft - 0.25" Oil Supply Line	N/A	$ 1.00	per ft	50	ft	$ 50.00
		20% Micelaneous Cost:				$ 18,097.80
		Total Cost of System for 10 years:				$ 108,586.80

- Assuming replacement every 2000 hours for 10 years. Based on estimates of usage from Research, this would require 2 replacements
- Manufacturing took an approximate guess at the total cost of the seals after reviewing the drawings briefly. This cost doesn't include the reconditioning costs every 2000 hours.
- Assumed the oil filters and pressurized air filters would have to be replaced twice a year for ten years
- The prices for the oil pump and oil reservoir are currently just “WAG”

Option 2 Cost Estimate

Radial Air Bearing / Thrust Air Pad Cost Summary							
Description	Part #	Cost		Quantity		Total	
40 x 80 Radial Air Bearing Pad	S3240W5001	$ 650.00	per pad	8	pads	$	5,200.00
200 mm Thrust Air Pad	S1020001	$ 860.00	per pad	1	pads	$	860.00
Modification Cost (In House)	N/A	$ 90.00	per hour	8	hours	$	720.00
End Ball Mounting Screw	S8020B01	$ 55.00	per unit	8	units	$	440.00
Stein Circumferential Seals		$ 1,000.00	per seal	4	seals	$	4,000.00
2 Stage Air Filter	S90L006	$ 225.00	per unit	1	units	$	225.00
Air Dryer	S90L007	$ 150.00	per unit	1	units	$	150.00
Air Regulator	S90R002	$ 90.00	per unit	1	units	$	90.00
Replacement Filter	S90L008	$ 150.00	2 per unit	20	units	$	3,000.00
Replacement Desiccant Dryer	S90L009	$ 30.00	per unit	20	units	$	600.00
1/4" NPT Hex Nipple	S90F091	$ 12.00	per unit	2	units	$	24.00
Straight Fittings	S90F036	$ 2.00	per unit	9	units	$	18.00
Right Angle Fittings	S90F044	$ 4.00	per unit	4	units	$	16.00
T-Fittings	S90F053	$ 5.00	per unit	4	units	$	20.00
100 ft - 0.25" Plastic Air Tubing	S90T004	$ 1.00	per ft	100	ft	$	100.00
		20% Micelaneous Cost:				$	3,092.60
		Total Cost of System for 10 years:				$	18,555.60

Air Bushing/ Thrust Air Pad Cost Summary							
Description	Part #	Cost		Quantity		Total	
2" Air Bushing	S305001	$ 420.00	per bearing	2	bearings	$	840.00
200 mm Thrust Air Pad	S1020001	$ 860.00	per pad	1	pads	$	860.00
Modification Cost (In House)	N/A	$ 90.00	per hour	8	hours	$	720.00
Stein Circumferential Seals		$ 1,000.00	per seal	4	seals	$	4,000.00
2 Stage Air Filter	S90L006	$ 225.00	per unit	1	units	$	225.00
Air Dryer	S90L007	$ 150.00	per unit	1	units	$	150.00
Air Regulator	S90R002	$ 90.00	per unit	1	units	$	90.00
Replacement Filter	S90L008	$ 150.00	2 per unit	20	units	$	3,000.00
Replacement Desiccant Dryer	S90L009	$ 30.00	per unit	20	units	$	600.00
1/4" NPT Hex Nipple	S90F091	$ 12.00	per unit	2	units	$	24.00
Straight Fittings	S90F036	$ 2.00	per unit	3	units	$	6.00
100 ft - 0.25" Plastic Air Tubing	S90T004	$ 1.00	per ft	100	ft	$	100.00
		20% Micelaneous Cost:				$	2,123.00
		Total Cost of System for 10 years:				$	12,738.00

All the air supply filters and desiccant dyers would be replaced twice a year for 10 years

The bearings would never have to be replaced in the 10 year life span.

The prices for the seals are just “WAG” right now.

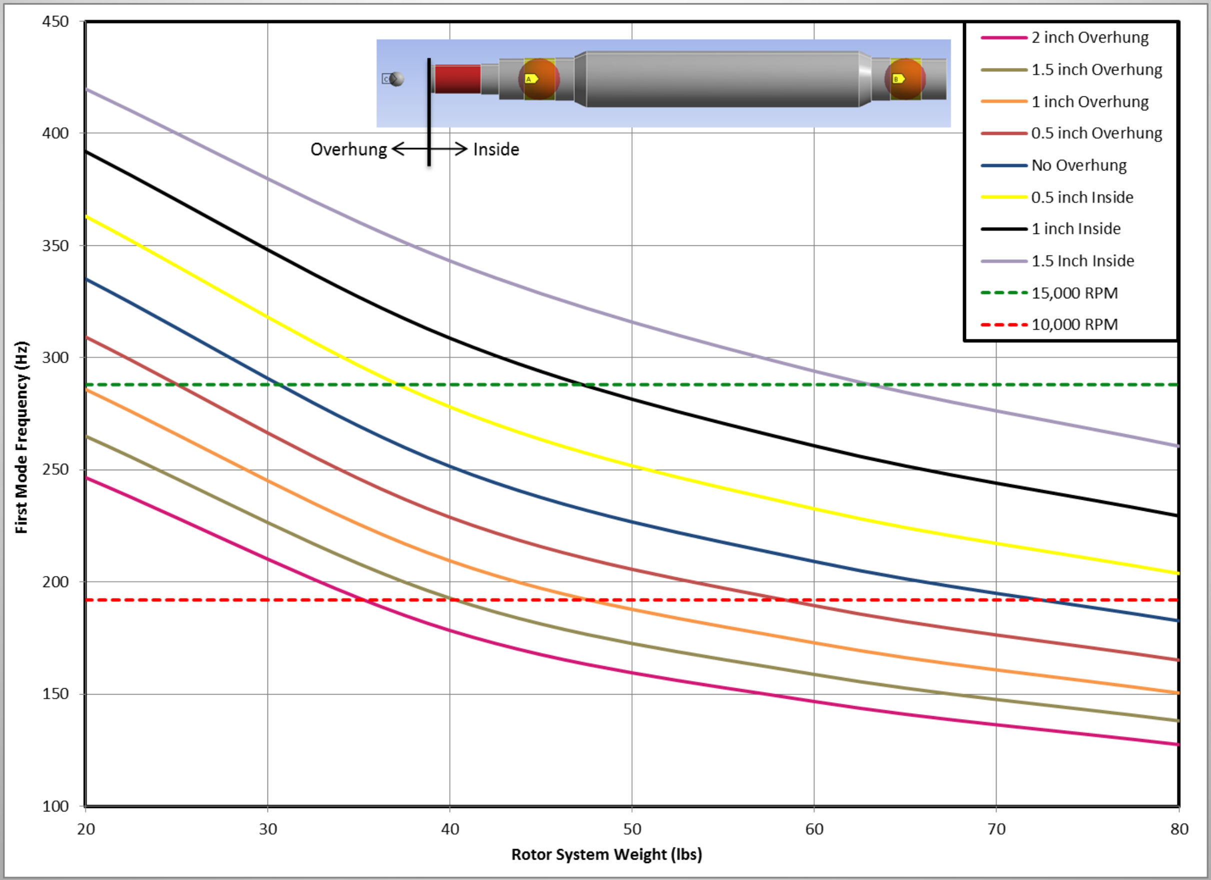
Option 3 Cost Estimate

Hydrodynamic Foil Bearings						
Description	Part #	Cost		Quantity		Total
Development Costs	N/A	N/A	N/A	N/A	N/A	$ 225,000.00
5.5" x 6" Radial Foil Bearing	N/A	$5,000.00	per bearing	2	bearings	$ 25,000.00
8.5" Thrust Bearing	N/A	$8,000.00	per bearing	1	bearings	$ 25,000.00
Stein Circumferential Seals	N/A	$1,000.00	per seal	4	seals	$ 4,000.00
2 Stage Air Filter	S90L006	$ 225.00	per unit	1	units	$ 225.00
Air Dryer	S90L007	$ 150.00	per unit	1	units	$ 150.00
Air Regulator	S90R002	$ 90.00	per unit	1	units	$ 90.00
Replacement Filter	S90L008	$ 150.00	2 per unit	20	units	$ 3,000.00
Replacement Desiccant Dryer	S90L009	$ 30.00	per unit	20	units	$ 600.00
1/4" NPT Hex Nipple	S90F091	$ 12.00	per unit	2	units	$ 24.00
100 ft - 0.25" Plastic Air Tubing	S90T004	$ 1.00	per ft	100	ft	$ 100.00
		20% Micelaneous Cost:				$ 56,637.80
		Total Cost of System for 10 years:				$ 339,826.80

- The prices for the bearings were supplied by a foil bearing manufacturer, it was not an exact quote, just an educated guess.
- The bearings would never have to be replaced in the 10 year life span.
- The air filters and desiccant dyers would have to be replaced twice a year for 10 years.
- The prices for the seals are just “WAG” right now.

Shaft Dynamics – Effects of Overhung Load

- Shaft currently has a rotor weight of 70 lbs of mass with the cg at about 0.25” inside the front edge
- The most efficient way to increase the stiffness of the shaft is to move the load further inward.
- Every option being considered currently is not able to operate above the first critical frequency of 288Hz. Decision on how to address this issue is not part of this study.



Option 1 - Summary

Areas of Confidence

- This system has been used in this application before and they have had relatively good success with it, so the confidence level is high.
- Enough oil should be able to be able to be fed through the stators depending on the stator design and preliminary estimates. It won't be cheap or simple but it can be done.

Areas of Concern:

- The 9'x15' wind tunnel has expressed that the system can leak. Their system is contained inside. ANCFII is not therefore an oil leak could cause severe problems with the environment.
- The bearings would require the most maintenance and downtime of any of the bearing options. Bearings would have to be changed out and seals reconditioned approximately every three years.
- These bearings are the least acoustically friendly bearing and would also generate the most heat of any of the other technologies.
- With all the mechanical seals and parts required, this system is more expensive than the hydrostatic bearing option.

Option 2 - Summary

Areas of Confidence

- This configuration would reduce some of the risk associated with a thrust overload. The laws of physics are helping out in this case, the higher the load gets, the smaller the bearing gap becomes which increases the bearing load capacity.
- The porous media in the bearing would act like a large break pad in the event of an air supply failure and slow the rig without allowing the shaft to seize and overload the rest of the drive line.
- Most economical bearing option of the three.
- Because there is no contacting surfaces, the noise generated, friction losses, and maintenance are all kept considerably lower than its alternatives.
- The supply air can be routed through the shaft and not the stators keeping the cost of testing in ANCFII lower than having to modify the stators for lubrication to be fed through.

Areas of Concern:

- We would have to customize the thrust bearing by fabricating a hole for the shaft to fit through, although the manufacturer has given okay that this would be acceptable.
- These bearings have never been used on an application like ours so prototype testing would have to be a requirement shortly after making the selection.

Option 3 - Summary

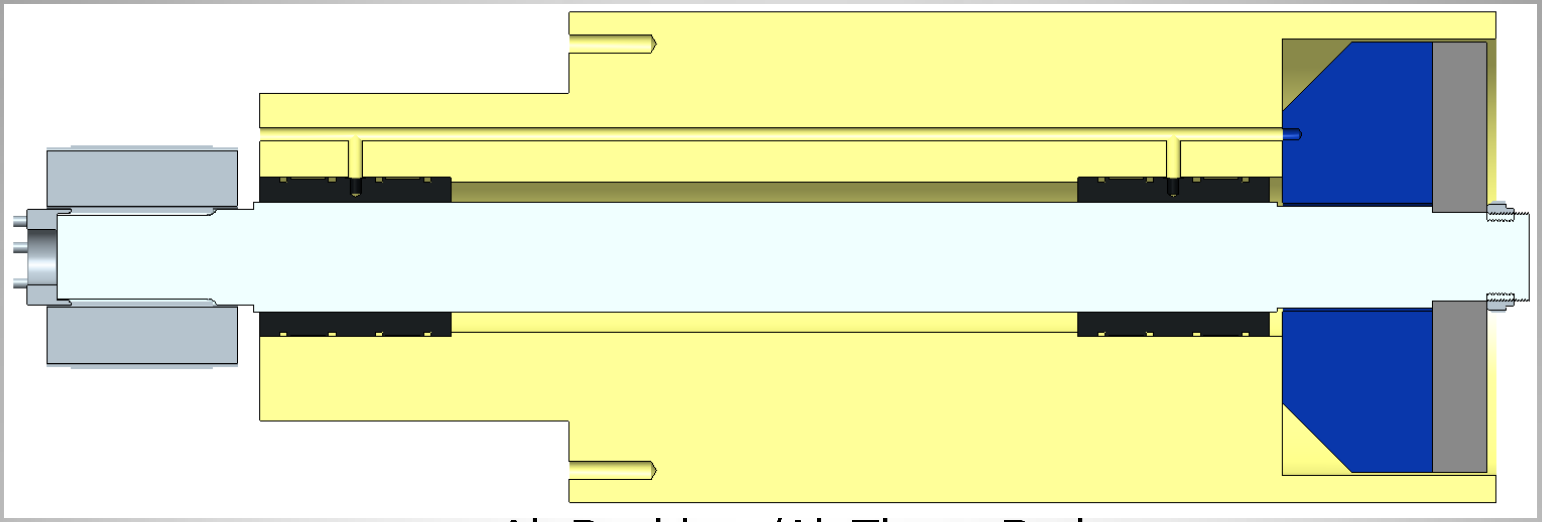
Areas of Confidence

- This bearing does not have any reasonable speed limitations.
- These bearings have been used by researchers at the center with a good record of success.
- They generate more load capacity the faster the shaft spins which is the exact physics of our test rig so they are well suited for the application.
- Shaft dynamics are more easily handled with this shaft because the diameter has to be so much larger than the other bearing options.

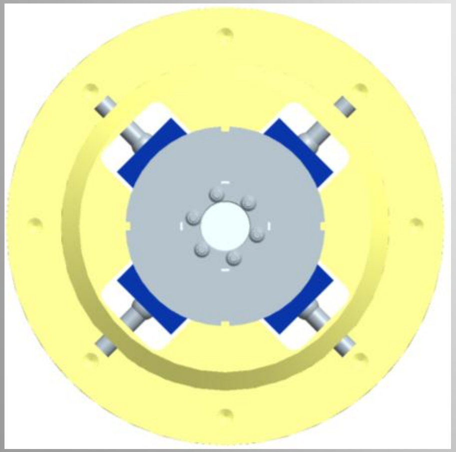
Areas of Concern:

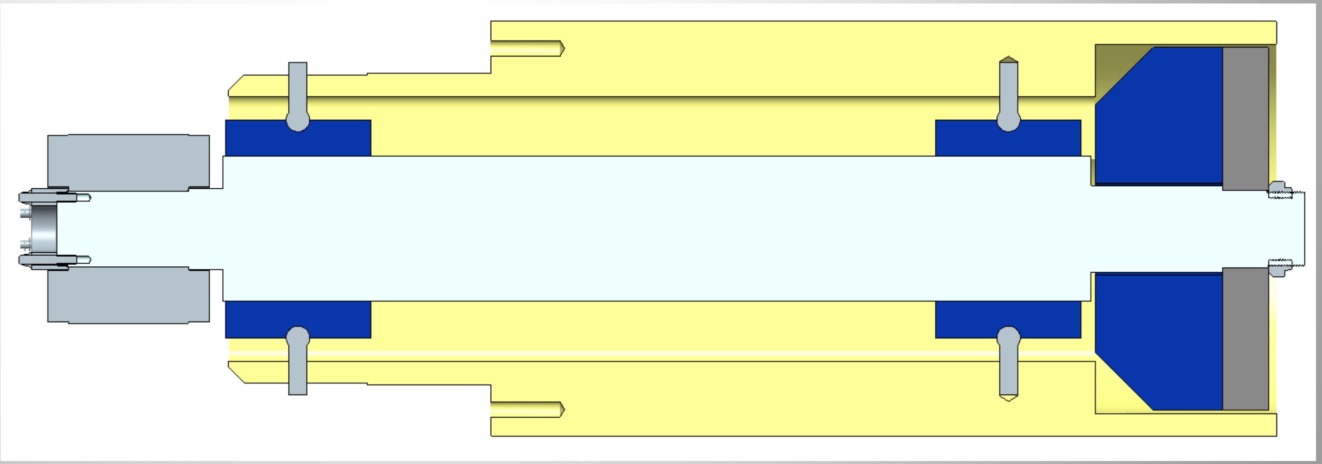
- The bearings would have to be custom bearings which drives the cost up well above other bearing technologies that have been investigated.
- They are supposed to be able to operate without air supply, but they would still require a bleed air system and the thrust bearing would require supplemental pressurized air which then makes it similar to the hydrostatic concept only a lot more expensive.
- The same air filtration requirements are required on this system as the hydrostatic system.

Selected Option Concepts



Air Bushings/Air Thrust Pad





Concave Radial Air Pads/Air Thrust Pad

Recommended Action

- Based on the data presented, I believe that Option 2 using the hydrostatic porous media air bearings is the best option.

  - ✓ Decision of which type of radial hydrostatic bearing to use is contingent on further development of the shaft dynamic issue.

  - ✓ This is contingent on the fact that prototype testing would need to be done to verify the manufacturer's claims and data.